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(71) Applicant: AISIN AW CO., LTD.
10, Takane
Fujilcho
Anjo-shi Aichi-ken (JP)

(72) Inventor: Hamajima, Tetsuo
1, Toyota-cho
Toyota-shi, Aichi-ken (JP)
Inventor: Oba, Hidehiro
1, Toyota-cho
Toyota-shi, Aichi-ken (JP)
Inventor: Kimura, Hiromichi
1, Toyota-cho
Toyota-shi, Aichi-ken (JP)
Inventor: Ando, Masahiko
10, Takane,

Fujilcho
Anjo-shi, Aichi-ken (JP)
Inventor: Fukatsu, Akira
10, Takane,
Fujilcho
Anjo-shi, Aichi-ken (JP)
Inventor: Yamamoto, Yoshihisa
10, Takane,
Fujilcho
Anjo-shi, Aichi-ken (JP)
Inventor: Hayabuchi, Masahiro
10, Takane,
Fujilcho
Anjo-shi, Aichi-ken (JP)
Inventor: Tsukamoto, Kazumasa
10, Takane,
Fujilcho
Anjo-shi, Aichi-ken (JP)

(74) Representative: VOSSIUS & PARTNER
Siebertstrasse 4
D-81675 München (DE)

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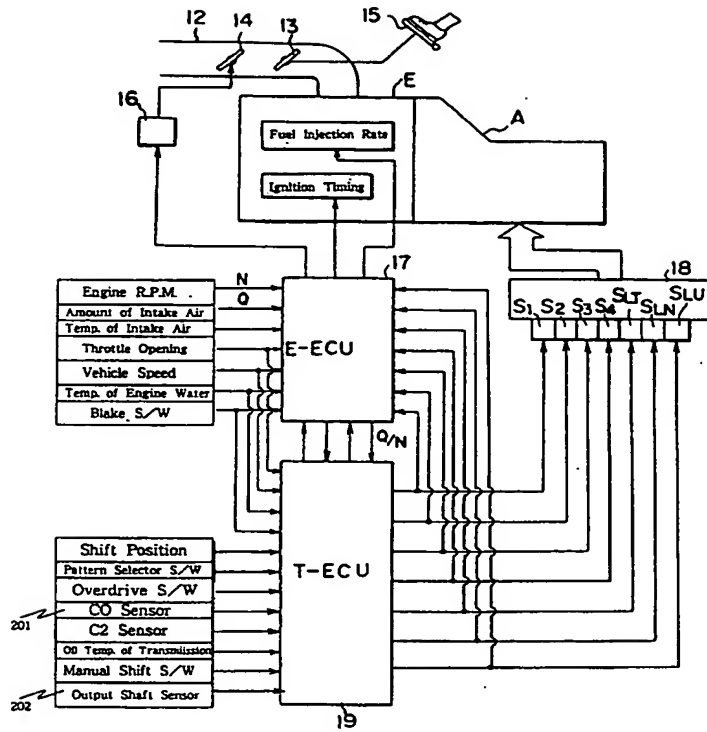
(54) Shift control system for automatic transmissions.

(57) A shift control system for shifting an automatic transmission from a predetermined gear stage to another by draining an oil pressure from predetermined frictional engagement means connected to an engine to release the same frictional engagement means and by feeding the oil pressure to another frictional engagement means to apply the same frictional engagement means while controlling the drain oil pressure of the frictional engagement means to be released and the application oil pressure of the frictional engagement means to be applied, during the shift. The shift control system comprises blowup detecting means for detecting the blown-up state of the engine being shifted in a clutch-to-clutch manner, and hydraulic control means for controlling at

least one of the drain oil pressure of the frictional engagement means to be released and the application oil pressure of the frictional engagement means to be applied, on the basis of the blown-up state of the engine detected by the blowup detecting means, such that the amount of blowup of the engine may fall within a predetermined range. Since the rotation changing state of the engine being shifted is reflected upon the hydraulic control of the frictional engagement means participating in the clutch-to-clutch shift, it is possible to reliably prevent any excessive blowup of the engine and the reduction of the output shaft torque due to the tie-up of the frictional engagement means at all times.

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Fig.2



The present invention relates to a system for controlling the shift of an automatic transmission and, more particularly, to a shift control system for an automatic transmission to execute the so-called "clutch-to-clutch shift".

The shift, which is effected by releasing first frictional engagement element and by applying second frictional engagement element, is called the "clutch-to-clutch shift". In this shift, the so-called "overlap state", in which the two frictional engagement elements are temporarily given a torque capacity higher than a predetermined value, is effected for an up-shift whereas the so-called "underlap state", in which the two frictional engagement elements are temporarily released, is effected for a downshift by the oil pressure which is controlled by electric signals.

At the time of the aforementioned shift in the prior art, the controlled oil pressure has its correction set to apply the amount of correction to subsequent and later shifts by detecting the blown-up state of the engine in terms of the input/output R.P.M. so as to prevent the excessive engine blowup, which is caused by the failure of the oil pressure to change, as expected, for the electric signals or the reduction of the output shaft torque due to the tie-up (as disclosed in U.S.P. No. 5,018,068).

In the construction described above, however, the correction of the controlled oil pressure is so set that the engine may not be blown up. Since the controlled oil pressure is not corrected when the blowup of the engine is not detected, the output shaft torque drops to invite a shift shock if the tie-up occurs.

It is, therefore, conceivable to detect the tie-up so that the correction of the controlled oil pressure may be set on the basis of the detected value. However, it is difficult to detect the tie-up from the input/output R.P.M. Still the worse, it costs a high price and enlarges the size of the system to arrange a special torque sensor for detecting the torque from the input/output shaft, for example. For this arrangement, the automatic transmission has to be substantially modified.

It is, therefore, an object of the present invention to solve the above-specified problems and to provide a shift control system capable of preventing any drop of the output shaft torque and any excessive blowup due to a tie-up in the clutch-to-clutch shift.

In order to achieve the above-specified object, according to the present invention, there is provided a shift control system for an automatic transmission, comprising: first and second frictional engagement elements; and first and second hydraulic servos for applying/releasing the first and second frictional engagement elements by feeding/draining

an oil pressure thereto, so that a predetermined gear stage is shifted to another gear stage by draining the oil pressure from the first hydraulic servo to release the first frictional engagement element and by feeding the oil pressure to the second hydraulic servo to apply the second frictional engagement element, while controlling the drain oil pressure from the first hydraulic servo and the feed oil pressure to the second hydraulic servo during the gear stage shift, wherein the improvement comprises: blowup detecting means for detecting the blown-up state of an engine during the shift; and hydraulic control means for comparing the engine blown-up state detected by the blowup detecting means and a target blown-up state, in which the engine is blown up to a predetermined extent from the non-blown-up state of the engine, to control at least one of the drain oil pressure and the application oil pressure thereby to effect the target blown-up state, such that the blown-up state of the engine is lowered, if it is higher than the target blown-up state, but is raised if it is lower than the target blown-up state.

The automatic transmission according to the present invention comprises: first frictional engagement means 1 including a first frictional engagement element and a first hydraulic servo for releasing the first frictional engagement element when an oil pressure is drained therefrom; and second frictional engagement means 2 including a second frictional engagement element and a second hydraulic servo for applying the second frictional engagement element when the oil pressure is fed thereto, so that the shift from the predetermined gear stage to another is executed by releasing the first frictional engagement element and applying the second frictional engagement element. At the time of this shift, the drain oil pressure and the feed oil pressure of the individual first and second hydraulic servos are controlled, but these controls are carried out on the basis of the blown-up state of the engine E. Specifically, at the time of that shift, the blown-up state of the engine E is detected by blowup detecting means 3, and at least one of the feed oil pressure and the drain oil pressure is controlled by hydraulic control means 4 on the basis of the blown-up state detected, so that the blown-up state of the engine may be set at all times from the non-blown-up state of the engine to a target blown-up state in which the engine is blown up to a predetermined extent. Specifically, at least one of the drain oil pressure and the feed oil pressure is corrected by setting the correction of the duty ratio to a zero value if the blown-up state of the engine is identical to the target blown-up state, to a positive value if the blown-up state of the engine is higher than the target blown-up state, and to a negative value if the blown-up state of the

engine is lower than the target blown-up state.

Fig. 1 is a block diagram showing the present invention conceptionally;

Fig. 2 is a block diagram showing a control system conceptionally according to one embodiment of the present invention;

Fig. 3 is a diagram showing a gear train of the automatic transmission mainly;

Fig. 4 is a diagram presenting an operation table for setting individual gear stages;

Fig. 5 is a diagram showing an array of range positions to be selected by a shift lever;

Fig. 6 is a diagram showing a portion of the hydraulic circuit;

Fig. 7 is a flow chart showing a control routine for learning/controlling the duty ratio on the basis of the blown-up state of the engine; and

Fig. 8 is a diagram plotting the relation between the blown-up state of the engine and the corrected value of the duty ratio.

One embodiment of the present invention will be described in the following with reference to the accompanying drawings. Fig. 2 is a diagram showing an entire control system of the embodiment of the present invention. An engine E connected to the automatic transmission A is equipped in its intake pipe 12 with a main throttle valve 13 and a sub-throttle valve 14 located upstream of the former. The main throttle valve 13 is so connected to an accelerator pedal 15 that it is controlled according to the depression of the accelerator pedal 15. On the other hand, the sub-throttle valve 14 is controlled by a motor 16. There is provided an engine electronic control unit (E-ECU) 17 for controlling the motor 16 to regulate the opening that sub-throttle valve 14 and for controlling the fuel injection rate and the ignition timing of the engine E. This electronic control unit 17 is composed mainly of a central processing unit (CPU), memory units (RAM and ROM) and an input/output interface and is fed as control data with a variety of signals including an engine (E/G) R.P.M. N, an amount Q of intake air, a temperature of intake air, a throttle opening, a vehicle speed, a temperature of engine water and a brake switch.

In the automatic transmission A, a hydraulic control unit 18 controls the shift and a lockup clutch. The hydraulic control unit 18 is so constructed as to be electrically controlled and is equipped with: first to third shift solenoid valves S1 to S3 for executing the shift; a fourth solenoid valve S4 for controlling an engine braking state; a linear solenoid valve SLT for controlling the line pressure; a linear solenoid valve SLN for controlling an accumulator back pressure; and a linear solenoid valve SLU for controlling the lockup clutch.

There is further provided an automatic transmission electronic control unit (T-ECU) 19 for con-

trolling the shift, the line pressure and/or the accumulator back pressure by outputting signals to those solenoid valves. This electronic control unit 19 is composed mainly of a central processing unit (CPU), memory units (RAM and ROM) and an input/output interface and is fed as the control data with the throttle opening, the vehicle speed, the engine water temperature, the brake switch signal, a shift position signal, a pattern select switch signal, an overdrive switch signal, a signal coming from a C0 sensor 201 for detecting the R.P.M. of a later-described clutch C0, a signal coming from an output shaft sensor 202 for detecting the R.P.M. of an output shaft 65, a signal coming from a C2 sensor for detecting the R.P.M. of a second clutch C2, an oil temperature of the automatic transmission, and a signal of a manual shift switch.

Moreover, the automatic transmission electronic control unit 19 and the engine electronic control unit 17 are connected with each other for data communications. Specifically, signals such as a signal of an amount (Q/N) of intake air per revolution are sent from the engine electronic control unit 17 to the automatic transmission electronic control unit 19, whereas signals such as a signal equivalent to a command signal for each solenoid valve or a signal for commanding a gear stage are sent from the automatic transmission electronic control unit 19 to the engine electronic control unit 17.

More specifically, the automatic transmission electronic control unit 19 decides the gear stage, the ON/OFF of the lockup clutch, or the regulated pressure level of the line pressure or the engaging pressure on the basis of the data inputted and the map stored in advance and outputs a command signal to a predetermined solenoid valve on the basis of the decision result to decide a failure or perform a control based on the decision. On the other hand, the engine electronic control unit 17 not only controls the fuel injection rate, the ignition timing and/or the opening of the sub-throttle valve 14 on the basis of the data inputted but also lowers the output torque temporarily by reducing the fuel injection rate at the shifting time of the automatic transmission A, by changing the ignition timing and/or by throttling the opening of the sub-throttle valve 14.

Fig. 3 is a diagram showing one embodiment of the gear train of the automatic transmission A described above. As shown, the gear train is constructed to set five forward and one reverse gear stages. Specifically, the automatic transmission A is constructed of a torque converter 20, an auxiliary transmission unit 21 and a main transmission unit 22. The torque converter 20 is equipped with a lockup clutch 23 which is interposed between a front cover 25 integrated with a pump impeller 24 and a member (or hub) 27 integrated with a turbine

runner 26. The engine has its crankshaft (although both are not shown) connected to the front cover 25, and an input shaft 28 having the turbine runner 26 connected thereto is connected to a carrier 30 of an overdrive planetary gear mechanism 28 constituting the auxiliary transmission unit 21.

Between the carrier 30 and a sun gear 31 of the planetary gear mechanism 29, there are interposed the multi-disc clutch C0 and a one-way clutch F0. Incidentally, this one-way clutch F0 is applied in case the sun gear 31 rotates forward (i.e., in the rotating direction of the input shaft 28) relative to the carrier 30. There is also provided a multi-disc brake B0 for braking the rotation of the sun gear 31 selectively. Moreover, a ring gear 32 acting as the output element of the auxiliary transmission unit 21 is connected to an intermediate shaft 33 acting as the input element of the main transmission unit 22.

In the auxiliary transmission unit 21, therefore, the planetary gear mechanism 29 rotates as a whole with the multi-disc clutch C0 or the one-way clutch F0 being applied, so that the intermediate shaft 33 rotates at the same speed as that of the input shaft 28, thus establishing a lower gear stage. With the brake B0 being applied to stop the rotation of the sun gear 31, moreover, the ring gear 32 is accelerated with respect to the input shaft 28 to establish a higher gear stage.

On the other hand, the main transmission unit 22 is equipped with three sets of planetary gear mechanisms 40, 50 and 60, which have their individual rotary elements connected, as follows. Specifically, a sun gear 41 of the first planetary mechanism 40 and a sun gear 51 of the second planetary mechanism 50 are integrally connected to each other. Moreover, a ring gear 43 of the first planetary mechanism 40, a carrier 52 of the second planetary mechanism 50 and a carrier 62 of the third planetary mechanism 60 are connected to one another, and an output shaft 65 is connected to the carrier 62 of the third planetary mechanism 60. In addition, the second planetary mechanism 50 has its ring gear 53 connected to a sun gear 61 of the third planetary mechanism 60.

The gear train of this main transmission unit 22 can set one reverse and four forward gear stages and is composed of the following clutches and brakes for that settings. Of these, the clutches will be described at first. A first clutch C1 is interposed between a ring gear 53 of the second planetary mechanism 50 and the sun gear 61 of the third planetary mechanism 60, which are connected to each other, and the intermediate shaft 33. Moreover, the second clutch C2 is interposed between the sun gear 41 of the first planetary mechanism 40 and the sun gear 51 of the second planetary mechanism 50, which are connected to each other,

and the intermediate shaft 33.

Here will be described the brakes. A first brake B1 is a band brake which is arranged to stop the rotations of the sun gears 41 and 51 of the first and second planetary mechanisms 40 and 50. Between these sun gears 41 and 51 (i.e., the common sun gear shaft) and a casing 66, there are arrayed in series a first one-way clutch F1 and a second brake B2 or a multi-disc brake. Of these, the first one-way clutch F1 is applied when the sun gears 41 and 51 are to rotate backward (of the rotating direction of the input shaft 28). A third brake B3 or a multi-disc brake is interposed between a carrier 42 of the first planetary mechanism 40 and the casing 66. Between a ring gear 63 of the third planetary mechanism 60 and the casing 66, there are arranged in parallel a fourth brake B4 or a multi-disc brake for braking the rotation of the ring gear 63 and a second one-way clutch F2. Incidentally, this second one-way clutch F2 is applied when the ring gear 63 is to rotate backward.

The automatic transmission A thus far described can set five forward and one reverse gear stages by applying/releasing the individual clutches and brakes, as tabulated in an operation table of Fig. 4. In Fig. 4: symbols ○ indicate the applied state; symbols ● indicate the applied state to be taken at the time of engine braking; symbols △ indicate the applied or released state; and blanks indicate the released state.

The individual gear stages appearing in Fig. 4 are set according to the running ranges which are selected by operating the (not-shown) shift lever, and the positions of those running ranges to be selected by the shift lever are arrayed, as shown in Fig. 5. Specifically, the array is made such that a parking (P) range position is followed by a reverse (R) range position and such that a neutral (N) range position is arranged subsequent to the R-range but oblique to the array of the P-range and the R-range. A drive (D) range position is arranged subsequent to the N-range in parallel with the aforementioned array direction of the P-range position and the R-range position. Moreover, a 4th speed range position is subsequently arranged at a right angle with respect to the array direction of the N-range and the D-range. In addition, a 3rd speed range position is arranged subsequent to the 4th speed range position in parallel with the aforementioned array direction of the N-range and the D-range. Moreover, a 2nd speed range position is arranged to have a relation similar to that which is taken by the N-range relative to the aforementioned R-range position. Finally, a low (L) range position is arranged to have a relation similar to that which is taken by the 4th speed range position relative to the D-range position.

Of these running ranges: the D-range can establish the five forward gear stages; the 4th speed range can establish the four forward gear stages excepting the 5th speed or the overdrive gear stage; the 3rd speed range can establish the gear stages to the 3rd speed; the 2nd speed range can establish the gear stages to the 2nd speed; and the L-range can establish only the first speed. In case, therefore, the shift lever is manually operated between the 3rd speed range position and the 2nd speed range position, there is established either an upshift from the 2nd speed to the 3rd speed or a downshift from the 3rd speed to the 2nd speed.

As enumerated in the operation table of Fig. 4, the shifts between the 2nd speed and the 3rd speed are effected by the clutch-to-clutch shifts in which both the application and release of the second brake B2 and the third brake B3 are changed together. In order to smoothen these shifts, a hydraulic circuit shown in Fig. 6 is incorporated into the aforementioned hydraulic control unit 18.

In Fig. 6, reference numeral 70 designates a 2-3 timing valve. This 2-3 timing valve 70 is formed with: an inlet port 73 communicating with a drain oil passage 72 of a 2-3 shift valve 71; a drain pressure input port 75 communicating with the drain oil passage 72 through an orifice 74; an input port 78 communicating with a supply oil passage 76 leading from the 2-3 shift valve 71 to the second brake B2 through an orifice 77; a signal port 79 to be fed with the signal pressure from the linear solenoid valve SLU for the lockup clutch; and a drain port 80. Moreover, the 2-3 timing valve 70 has its spool 81 formed with: a land 82 positioned at its one end for opening/closing the drain port 80; a land 83 positioned at an intermediate portion while partitioning the drain pressure input port 75 and the inlet port 73 and formed with a drain pressure receiving face at the side of the drain pressure input port 75; and a radially reduced land 84 positioned at the other end while partitioning the input port 78 and the drain pressure input port 75 and formed with a supply pressure receiving face. Still moreover, the land 82 at one end abuts against a pressure receiving piston 86 through a spring 85, and the pressure receiving piston 86 is formed with a face for receiving the signal pressure coming from the signal port 79.

An accumulator 87 for the third brake B3 is connected through an orifice 89 to an oil passage 88 leading to the third brake B3 and has its back pressure controlled to control the applying pressure of the third brake B3 by the signal pressure coming from the aforementioned linear solenoid valve SLN. Moreover, an accumulator 90 for the second brake B2 likewise has its back pressure controlled to control the applying pressure of the second brake B2 by the signal pressure coming from the linear

solenoid valve SLN.

On the other hand, reference numeral 91 appearing in Fig. 6 designates a B-2 orifice control valve constituting fast fill means for the second brake B2. This B-2 orifice control valve 91 is formed with a control port 94 at its end portion opposed to the end portion which is equipped with a spring 93 for urging a spool 92 in the axial direction. The control port 94 is given communication with the second brake B2 through an orifice 95. The B-2 orifice control valve 91 is formed at its intermediate portion with an input port 96 connected to the aforementioned supply oil passage 76, and a second brake port 97 which is given communication with the input port 96 or not by the spool 92 and connected to the second brake B2. Further formed are a third brake port 99, which is connected to the third brake B3 through a later-described B-3 control valve 98, and a drain port 100 which is given communication with the third brake port 99 or not by the spool 92. At the end portion arranged with the spring 93, there is further formed a signal port 101 which is fed with the signal pressure from the third solenoid valve S3.

Here will be described the B-3 control valve 98. This is a valve for controlling the rate of applying/releasing the oil pressure of the third brake B3. This B-3 control valve 98 is formed with: a signal port 102 connected to the oil passage 88 for causing the oil pressure of the third brake B3 to act as the signal pressure; a D-port 104 connected to the 2-3 shift valve 71 through an orifice 103; a brake port 105 given communication with the D-port 104 or not and connected to the oil passage 88; and a brake port 106 given communication with the brake port 105 or not and connected to the third brake port 99 in the aforementioned B-2 orifice control valve 91. A spool 107 for opening/closing those ports has its one end portion caused to abut against a piston 109 through a spring 108. The B-3 control valve 98 is further formed with a control port 110, which is opened between the spool 107 and the piston 109 and fed with the 3rd speed pressure, and a signal pressure port 111 for applying the signal pressure of the linear solenoid valve SLU for the lockup clutch to the piston 109. Incidentally, reference numeral 112 appearing in Fig. 6 designates a 1-2 shift valve, and numeral 113 designates an orifice formed in the drain oil passage 74.

In the hydraulic control unit 18 having the aforementioned hydraulic circuit incorporated therinto, the 2-3 timing valve 70 and the B-3 control valve 98 are fed as their control pressure with the signal pressure of the linear solenoid valve SLU for the lockup clutch, so that the timing for applying/releasing the oil pressure of the third brake B3 can be controlled by that linear solenoid

valve SLU. Specifically, this control is carried out by controlling the duty ratio of the linear solenoid valve SLU by the electronic control unit 19 such that the duty ratio is controlled on the basis of the blown-up state of the engine E. An example for this control routine is shown in Fig. 7.

This example shown in Fig. 7 is directed to the case of an upshift from the 2nd to 3rd speeds. First of all, at step 1, it is decided whether or not that upshift has been outputted. If this shift is outputted, it is decided at step 2 whether or not the upshift is along the shift curve. If this shift curve is followed, it is decided (at step 3) whether or not the oil pressure exceeds a preset constant value, and (at step 4) whether or not the failure of the aforementioned linear solenoid valve SLU has been decided. In other words, step 3 is executed because the automatic transmission A has to be in a stable state, and step 4 is executed because the normal operation of the linear solenoid valve SLU is premised by the overlap control.

Next, the maximum of the blowup of the engine E till the inertial phase is determined. Specifically, it is decided at step 5 whether or not the difference $\Delta N (= N_{en} - N_{en-1})$ between the R.P.M. N_{en} of the engine E obtained at this time and the R.P.M. N_{en-1} of the engine E obtained at the previous time is larger than the maximum ΔN_{max} obtained already. If the decision result is "YES", the R.P.M. difference ΔN is latched at the maximum (at step 6), and it is then decided (at step 7) whether or not the inertial phase has started. If this inertial phase has not started, the routine is returned to upstream of step 5, and it is decided again whether or not the difference between the engine R.P.M. detected at the previous time and the engine R.P.M. detected at this time exceeds the maximum latched already. If this decision result is "NO", the routine skips step 6 to step 8. In other words, the maximum of the rotational fluctuations of the engine E, i.e., the maximum of the amount of blowup till the start of the inertial phase is determined by repeating the operations of step 5 and step 6 till the inertial phase starts.

If the inertial phase has started, it is decided (at step 8) whether or not no throttle change over a preset value has occurred during the shift. If this decision result is "YES", comparison is made between the aforementioned maximum of the blowup of the engine E and a target amount of blowup, in which the engine E is blown up to a predetermined extent from the non-blown-up state of the engine. If the maximum of the blowup of the engine E exceeds the aforementioned target amount of blowup, a correction value $\Delta Dslu$ is added (at step 9) to the duty ratio $Dslu$ which has already been latched for controlling the aforementioned linear solenoid valve SLU. This correction value $\Delta Dslu$ is set

according to the maximum blowup ΔN_{max} in the inertial phase and is plotted by way of example in Fig. 8. Specifically, in case the blowup of the engine E is high, the duty ratio $Dslu$ is augmented by increasing the correction value $\Delta Dslu$, so that the degree of overlap between the second brake B2 and the third brake B3 may be increased.

If the maximum blowup of the engine E is lower than the target blowup, the correction value $\Delta Dslu$ is set to a negative value to lower the duty ratio of the aforementioned linear solenoid valve SLU. In short, the control is made for an underlap tendency.

If, on the other hand, the aforementioned decision results of steps 1 to 4 and step 8 are "NO", no special control is made to return the procedure.

In the control unit thus far described, therefore, the overlap state or the underlap state between the frictional engagement means to take part in the clutch-to-clutch shift is controlled either while detecting the blown-up state of the engine E or according to the detected result, so that the amount of blowup of the engine E can be confined within a target range without being influenced by the aging of the hydraulic circuit or the like.

Incidentally, in the foregoing embodiment, the R.P.M. difference of the engine is used as the blowup amount of the engine, but this engine blowup may be determined from the difference between the C0 R.P.M. detected by the C0 sensor and the product of the R.P.M. of the output shaft detected from the output shaft sensor 202 and the gear ratio of the 2nd speed. Moreover, the embodiment is exemplified by a learning control in which the duty ratio is updated on the basis of the blowup amount till the start of the inertial phase. In the present invention, however, the duty ratio may be controlled on real time. Still moreover, the present invention should not be limited to the overlap control for the clutch-to-clutch upshift but can be applied to an underlap control for a clutch-to-clutch downshift. In the foregoing embodiment, furthermore, the description has been made upon the automatic transmission having the gear train shown in Fig. 3. However, the present invention should not be limited to that embodiment but can also be applied to a control unit for an automatic transmission having a gear train of another construction, so that it can be applied to an overlap control for a clutch-to-clutch upshift other than the upshift from the 2nd to 3rd speeds. In the present invention, furthermore, it is quite natural that the hydraulic circuit for the overlap control should not be limited to that having the construction shown in Fig. 5.

As has been described hereinbefore, according to the shift control system of the present invention, the oil pressure of at least one of the releasing oil pressure and the applying oil pressure is subjected

to such a learning control by correcting the duty ratio that the blown-up state of the engine for the clutch-to-clutch shift may always be changed from the non-blown-up state of the engine to the state of a predetermined target blowup. As a result, it is possible to reliably prevent the drop of the output shaft torque, which might otherwise be caused by the tie-up of the two frictional engagement elements due to the aging of the control device such as the hydraulic circuit, and to prevent any excessive blowup of the engine.

Claims

1. A shift control system for an automatic transmission, comprising: first and second frictional engagement elements; and first and second hydraulic servos for applying/releasing said first and second frictional engagement elements by feeding/draining an oil pressure thereto, so that a predetermined gear stage is shifted to another gear stage by draining the oil pressure from said first hydraulic servo to release said first frictional engagement element and by feeding the oil pressure to said second hydraulic servo to apply said second frictional engagement element, while controlling the drain oil pressure from said first hydraulic servo and the feed oil pressure to said second hydraulic servo during said gear stage shift, wherein the improvement comprises: blowup detecting means for detecting the blown-up state of an engine during said shift; and hydraulic control means for comparing the engine blown-up state detected by said blowup detecting means and a target blown-up state, in which the engine is blown up to a predetermined extent from the non-blown-up state of the engine, to control at least one of said drain oil pressure and said application oil pressure thereby to effect said target blown-up state, such that the blown-up state of the engine is lowered, if it is higher than said target blown-up state, but is raised if it is lower than said target blown-up state.
2. A shift control system according to claim 1, wherein said hydraulic control means includes: a pressure regulator valve for regulating said oil pressure and controlling the regulated value; and an electromagnetic solenoid valve adapted to be controlled by a duty ratio for changing the regulated pressure value of said pressure regulator valve, so that the engine blown-up state may be changed to said target blown-up state by changing said duty ratio.
3. A shift control system according to claim 2, wherein said hydraulic control means includes correction means for correcting said duty ratio to a zero value, if said engine blown-up state is identical to said target blown-up state, to a positive value, if said engine blown-up state is higher than said target blown-up state, and to a negative value if said engine blown-up state is lower than said target blown-up state.
4. A shift control system according to claim 2 or 3, wherein said pressure regulator valve regulates the oil pressure to be drained from said first hydraulic servo.

Fig.1

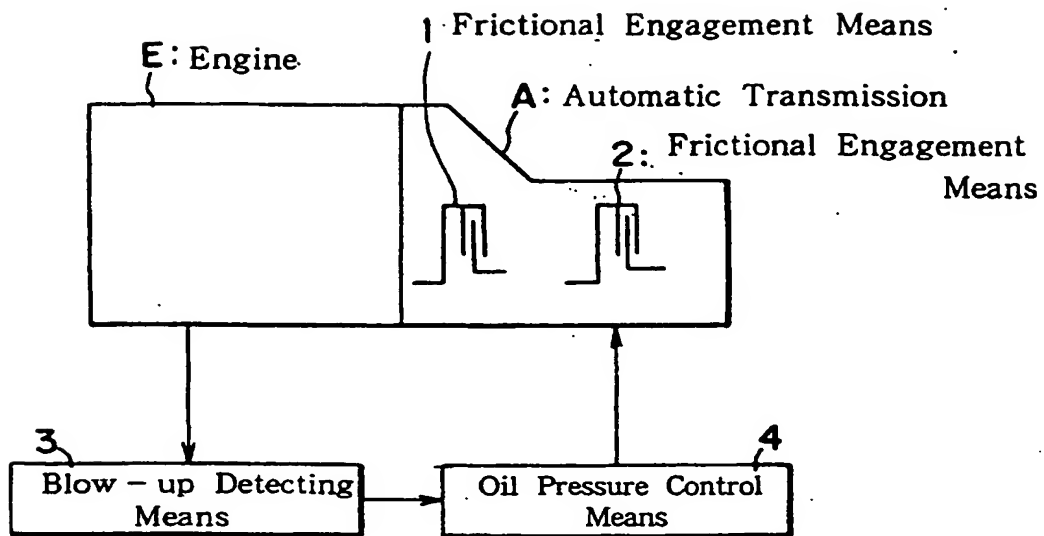


Fig.5

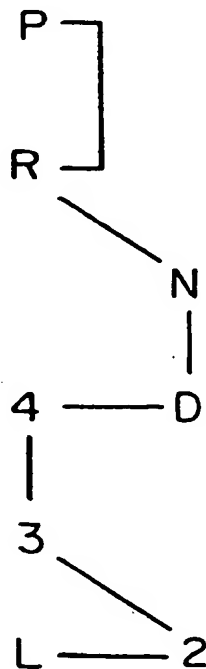


Fig.2

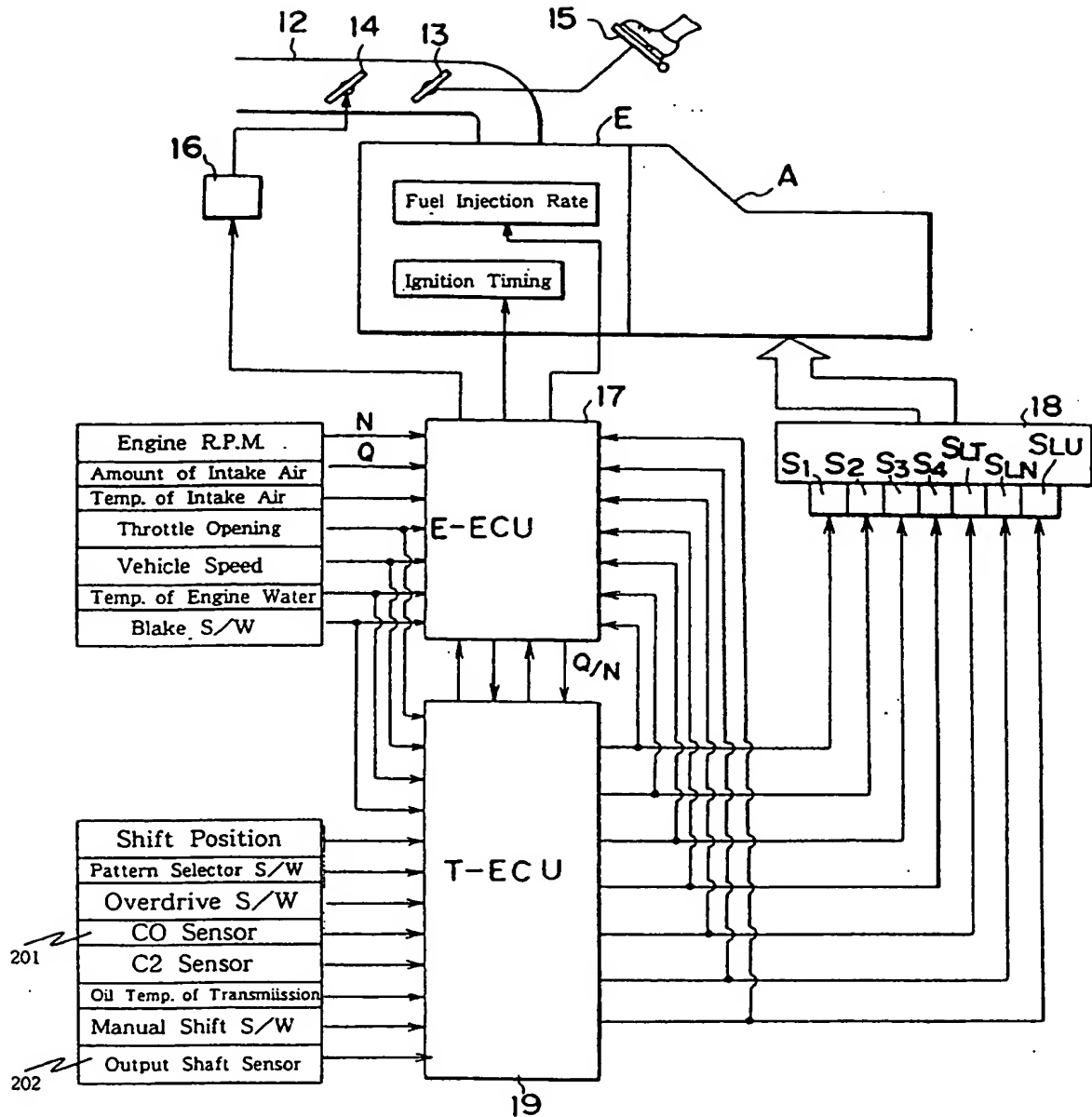


Fig.3

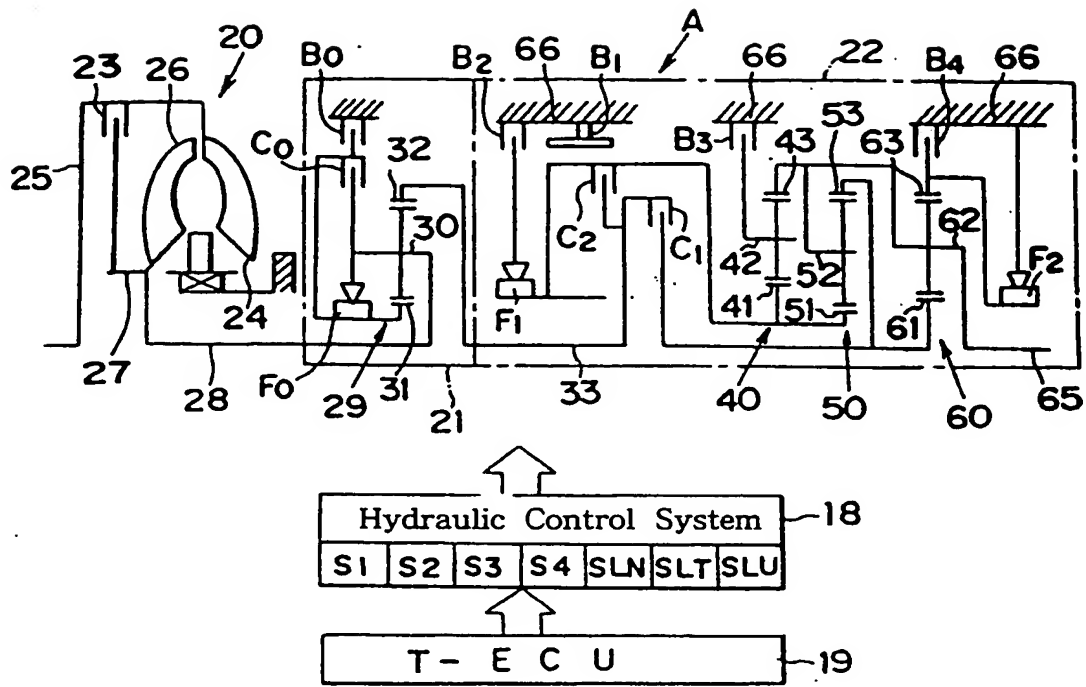


Fig.4

	C ₀	C ₁	C ₂	B ₀	B ₁	B ₂	B ₃	B ₄	F ₀	F ₁	F ₂
N				○							
Rev			○	○				○			
1st	○	○						●	○		○
2nd	●	○					○		○		
3rd	○	○			●	○			○	○	
4th	○	○	○			△			○		
5th		○	○	○		△					

Fig.6

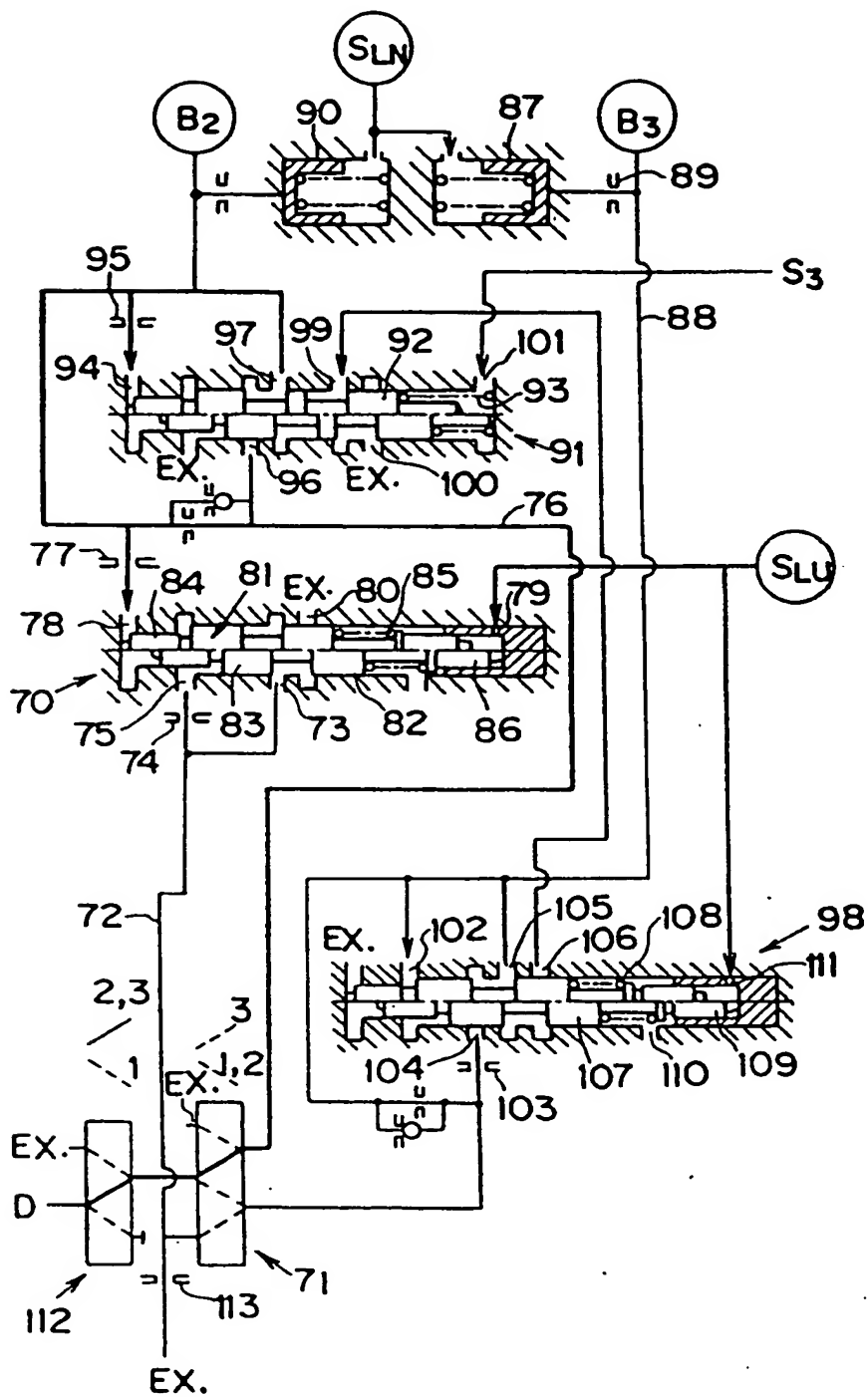


Fig.7

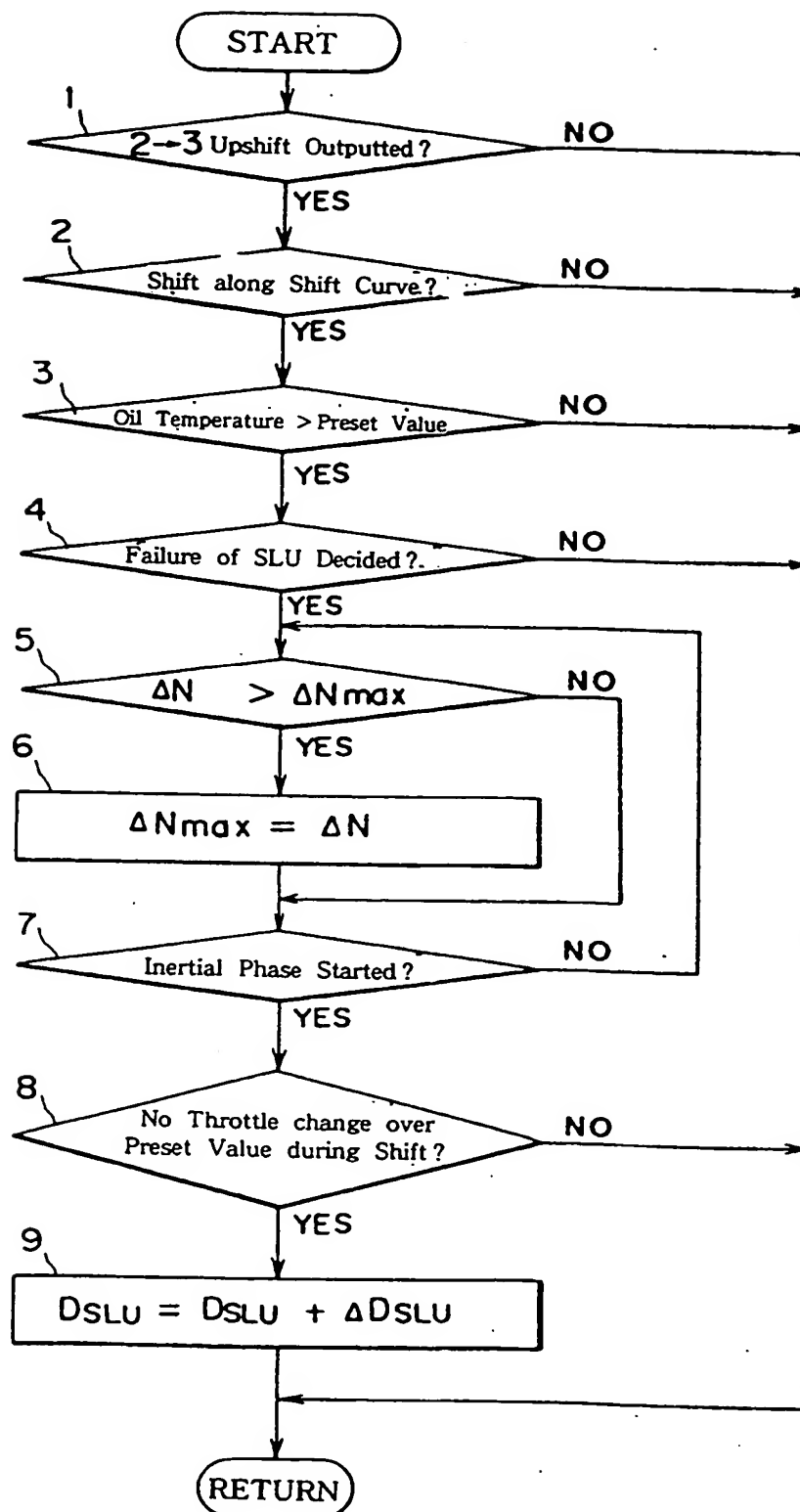
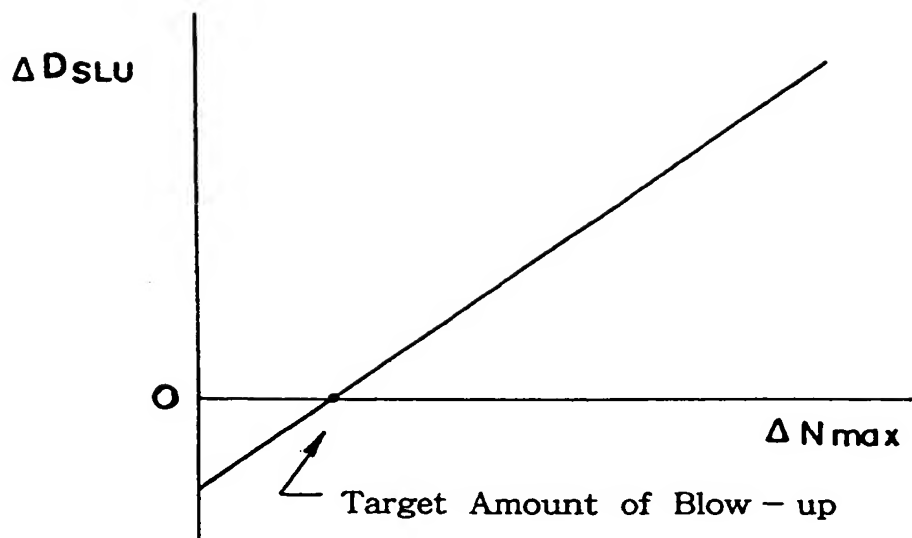


Fig.8



(19)



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(71) Applicant: AISIN AW CO., LTD.
Anjo-shi Aichi-ken (JP)

(72) Inventors:
• Hamajima, Tetsuo
Toyota-shi, Aichi-ken (JP)
• Oba, Hidehiro
Toyota-shi, Aichi-ken (JP)
• Kimura, Hiromichi
Toyota-shi, Aichi-ken (JP)

• Ando, Masahiko
Anjo-shi, Aichi-ken (JP)
• Fukatsu, Akira
Anjo-shi, Aichi-ken (JP)
• Yamamoto, Yoshihisa
Anjo-shi, Aichi-ken (JP)
• Hayabuchi, Masahiro
Anjo-shi, Aichi-ken (JP)
• Tsukamoto, Kazumasa
Anjo-shi, Aichi-ken (JP)

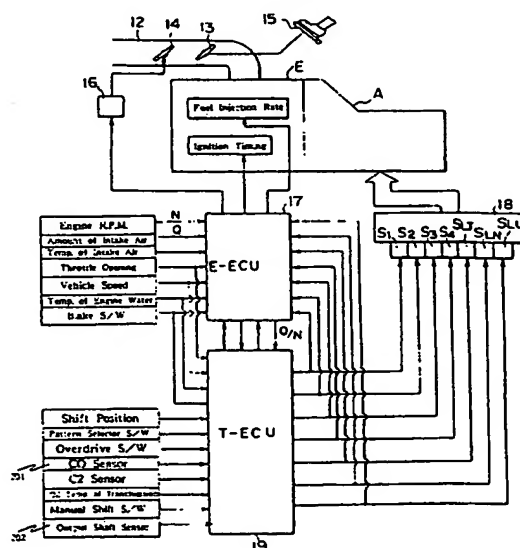
(74) Representative: VOSSIUS & PARTNER
Siebertstrasse 4
81675 München (DE)

(54) Shift control system for automatic transmissions

(57) A shift control system for shifting an automatic transmission from a predetermined gear stage to another by draining an oil pressure from predetermined frictional engagement means connected to an engine to release the same frictional engagement means and by feeding the oil pressure to another frictional engagement means to apply the same frictional engagement means while controlling the drain oil pressure of the frictional engagement means to be released and the application oil pressure of the frictional engagement means to be applied, during the shift. The shift control system comprises blowup detecting means for detecting the blown-up state of the engine being shifted in a clutch-to-clutch manner, and hydraulic control means for controlling at least one of the drain oil pressure of the frictional engagement means to be released and the application oil pressure of the frictional engagement means to be applied, on the basis of the blown-up state of the engine detected by the blowup detecting means, such that the amount of blowup of the engine may fall within a predetermined range. Since the rotation changing state of the engine being shifted is reflected upon the hydraulic control of the frictional engagement means participating in the clutch-to-clutch shift, it is possible to reliably prevent any excessive blowup of the engine and the reduction of the

output shaft torque due to the tie-up of the frictional engagement means at all times.

Fig.2



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DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.5)
A	US-A-5 003 842 (HATTA KATUHIRO ET AL) 2 April 1991 * the whole document *	1-4	F16H61/06 B60K41/08
P,A	EP-A-0 583 954 (MITSUBISHI MOTORS CORP) 23 February 1994 * abstract *	1-3	
A	PATENT ABSTRACTS OF JAPAN vol. 014, no. 193 (M-0963), 19 April 1990 & JP-A-02 037128 (MAZDA MOTOR CORP), 7 February 1990, * abstract *	1,3	
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A	EP-A-0 435 377 (GEN MOTORS CORP) 3 July 1991 * column 6, line 39 - column 7, line 15; figure 4 *	1	
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The present search report has been drawn up for all claims			
Place of search BERLIN		Date of completion of the search 22 February 1996	Examiner Gertig, I
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P, A	EP-A-0 578 223 (DEERE & CO) 12 January 1994 -----		
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The present search report has been drawn up for all claims			
Place of search BERLIN		Date of completion of the search 22 February 1996	Examiner Gertig, I
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